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FILM COOLING HEAT TRANSFER WITH HIGH FREE STREAM TURBULENCE

John J. Schauer
David J. Pestian

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FILM COOLING HEAT TRANSFER WITH HIGH FREE STREAM TURBULENCE

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ABSTRACT

Heat transfer coefficients are shown as functions of downstream and transverse plate coordinates as determined experimentally for different blowing ratios near one and for levels of free stream turbulence from 1 percent to 17 percent of the free stream velocity. Heat transfer between the plate and the free stream is reduced by the film cooling at higher levels of free stream turbulence. Heat transfer is increased on the plate in the region between the film cooling jets at all levels of blowing ratio for one percent free stream turbulence. This undesirable augmentation of the heat transfer is modeled analytically and related to the axial vortices generated near the film cooling jet inlets. The strength of these vortices and the subsequent heat transfer increases are related to the blowing ratio.

Nomenclature

A_s	side area under the cooling flow (m^2)
d	film cooling hole diameter (1.9cm)
M	blowing ratio ($\rho_{fc}U_{fc}/\rho_{fs}U_{fc}$)
T	static temperature (deg. K)
T*	local temperature difference from free stream value(deg. K)
T_m	difference between maximum temperature on a constant x/d plane and the associated free stream value (deg. K)
Tu	turbulence intensity (u'/u) (%)
c	gas specific heat at constant pressure (J/kg deg. K)
h	convection coefficient, q"/(T_w-T_{fs}) (W/m^{2}K)
h_{aw}	convection coefficient, q"/(T_{aw}- T_{fs}) (W/m^{2}K)
k	empirical constant in equation (2)
q"	local heat flux (W/m^2)
r	radial distance from center of axisymmetric coolant jet (m)
u	mean local stream wise velocity (m/s)
u'	fluctuating stream wise velocity component (m/s)
v	mean local vertical velocity (m/s)
w	mean local transverse velocity (m/s)
W	velocity related to the vortex strength (m/s)
x	stream wise dist. from downstream lip of injection hole (m)
y	vertical distance from injection surface (m)
z	span wise distance from centerline of injection hole (m)
δ	boundary layer thickness (m)
ρ	fluid density (kg/m^3)
γ	angle between cooling jet and the plate (radians)

Subscripts:

aw	evaluated at y=0 (wall value), with q"=0
fc	in the film cooling fluid
fs	in the free stream fluid
nofc	no film cooling
std	standard
w	evaluated at y=0 (wall value)

INTRODUCTION

Turbine blades in modern aircraft operate with the flow over them hot enough to melt the blades. Cooling is a necessity and the cooling flow can at most reach the blade temperature. Since the cooling flow is cooler than the main flow, it is available for film cooling after it has completed the blade cooling function. This experimental and analytical study shows the cooling potential at blowing ratios near one and at levels of Free Stream Turbulence (FST) up to 17 percent. The analytical study focuses particularly on the conditions where heat transfer is augmented rather than having the desirable heat transfer reducing effect.

This study utilizes the apparatus configured by Bons et al. (1994) and also used in the jet mixing study of Schauer and Bons (1994). Experimental data were collected with flow parameters in the range of typical turbine applications. The Reynolds number based on film cooling hole diameter was 20,000 and the blowing ratios were near one. The data presented corresponds to x/d from 0 to 60. The density ratio between the cooling flow and the main stream was approximately 0.93.

EXPERIMENTAL FACILITY

The wind tunnel used for the experiments has a 0.38m (width) by 0.18m (height) cross-section. An elliptical-edge bleed with vacuum suction is located 12.07cm upstream of the downstream lip of the film cooling injection hole (designated as x/d = 0). A 1.59mm diameter steel rod located 2.54cm from the elliptical-edge bleed is used to trip the new boundary layer and insure a span wise uniform turbulent boundary layer profile at the injection point. Without employing turbulence generating devices, the tunnel's free stream turbulence level is 0.9% (±0.05) and velocity uniformity is within ±2.5%. Free stream turbulence generation is accomplished by two methods for the present experimental data. A secondary flow is injected from two opposing rows of holes located on the top and

bottom of the wind tunnel 1.02m upstream of the boundary layer bleed at a velocity ratio (jet to free stream velocity) of 14 to produce a turbulence level of 17% (±0.85) at the film cooling injection station. To provide a turbulence level of 6.5% (±0.3), a standard square grid is installed 0.94m upstream of the coolant injection point. The film cooling is injected through a 1.9cm diameter, 35 degree inclined hole centered in the test section. The injection pipe length from the coolant access plenum to the exit is 3.5 hole diameters. A diagram of this experimental configuration is show in Schauer and Bons (2).

The data presented in this report were taken using a 4μm diameter tungsten hot wire. The hot wire and a flow temperature thermocouple (0.33mm bead diameter) are mounted on a vertical traverse. A velocity map of the injection plane (x-z plane at y = 0) was used to calibrate an orifice plate used to determine the film cooling flow rate. This flow rate and the local free stream density-velocity product are used to calculate the film cooling blowing ratio. The test surface downstream of the film cooling injection point is a constant heat flux film over a 10 cm layer of urethane foam. Heat transfer coefficients are based on the known heat flux per unit area and the temperature difference between the free stream and the local wall temperature as measured by axially spaced thermocouples in the plate. The cooling holes were moved sideways relatively to the plate to permit heat transfer coefficients to be determined as functions of both axial and transverse positions. Uncertainty in the heat transfer coefficients is estimated at ±4% but the relative values are much more accurate. Uncertainty in the velocity measurement, estimated at ±2.0%, is attributed to the calibration fit accuracy and the horizontal displacement between the probe and the flow thermocouple. The uncertainty in the temperature ratio data is ±0.7%.

ANALYTICAL DEVELOPMENT

At a blowing ratio near one, the cooling flow spreads and mixes slowly in the transverse direction at low levels of FST. Under these circumstances the axial vortex generated by the main stream flow sweeping under the cooling jet flow near the jet exit causes significant changes in the boundary layer where the vortex pair meet midway between holes.

The vortex strength is analyzed using the moment of momentum control volume approach from Shapiro (1953) Eq. (1.17) applied to the control volume L*L*L shown in Fig. 1.
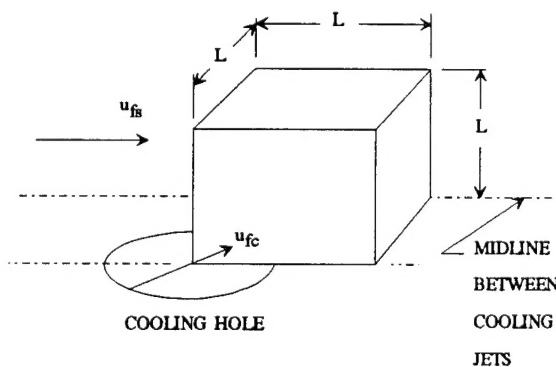

Figure 1. Control Volume for Vorticity Analysis

As shown in Fig. 1, L is the distance from the hole centerline to the midpoint between holes, in this case 1.5 d = L. The transverse component of the force on this control volume is associated with the reduced pressure under the cooling jet. This force was computed as

$$F = \int \Delta P dA_s \qquad (1)$$

where A_s is the area between the jet centerline and the plate and ΔP is the difference between the pressure in the main stream and the average pressure under the jet. This pressure difference was taken to be

$$\Delta P = 1/2\ k\rho u_{fs}2\gamma \qquad (2)$$

Here the combined effect of A_s, k, and γ are functions of the blowing ratio determined empirically. These empirical values will effect W_2 from Eq. 10, which is chosen based on the heat transfer data.

With the approximation that the force in the z direction given by Eq. (1) is located near the plate since the pressure differential is under the cooling flow, the lever arm about the top of the control volume (axis z = 0, y = L, extending in the x direction) is L. Then moment of momentum equation reduces to

$$LF = \rho u_{fs} L^2 (V \times r)_{out} \qquad (3)$$

Shapiro (1953) relates angular velocity (page 270) to this moment of momentum, giving

$$2\ \omega L^2 = (V x r)_{out} \qquad (4)$$

Estimating the velocity at the edge of the vortex, W, as the angular velocity ω times L and combining the above equations gives

$$W = L\ \omega = A_s\ k\ u_{fs}/4/L^2 \qquad (5)$$

In order to analyze heat transfer, the boundary layer development is investigated with the velocity corresponding to the vortex considered. The ratio of heat transfer coefficient with angular velocity to the heat transfer without (standard flat plate) is predicted.

Boundary layer growth at the midline between holes is examined using the control volume shown in Fig. 2. Notice that this control volume has a downward velocity v(δ) coming uniformly in the top and also the same velocity W coming out one side. The other side is on the midline and has no mean z component of velocity from symmetry.
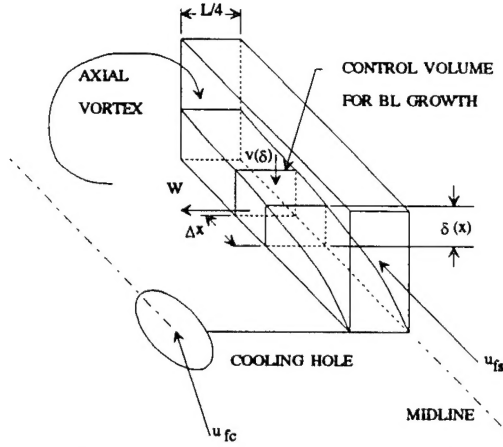
Figure 2. Control Volume for Boundary Layer Growth Near Midline

The law of conservation of mass for this control volume results in

$$\frac{L}{4}\frac{d}{dx}\int_0^\delta u\,dy + W\int_0^\delta dy - \frac{L}{4}v(\delta) = 0 \qquad (6)$$

The W in the second term of Eq. 6 represents the z component of velocity assumed constant from $y = 0$ to δ. The $v(\delta)$ in the third term is the vertical downward velocity component. Applying control volume x momentum to the same control volume and assuming no pressure changes in the x direction gives

$$-\frac{\tau_o}{\rho}\frac{L}{4} = \frac{L}{4}\frac{d}{dx}\int_0^\delta u^2 dy - u_{fs}v(\delta)\frac{L}{4} + W\int_0^\delta u\,dy \qquad (7)$$

Eliminating $v(\delta)$ between Eq. (6) and Eq. (7) relating the wall shear to δ via the Blausius relationship Eq. (10-18) of Kays and Crawford (1980) gives

$$-0.0225u_{fs}^2\left(\delta u_{fs}/\nu\right)^{-1/4}\frac{L}{4} = \frac{L}{4}u_{fs}^2\frac{d\delta}{dx}\int_0^1\left(\frac{u}{u_{fs}}\right)^2 d\left(\frac{y}{\delta}\right) -$$

$$u_{fs}W\delta - \frac{L}{4}u_{fs}\frac{d\delta}{dx}\int_0^1\frac{u}{u_{fs}}d\left(\frac{y}{\delta}\right) + u_{fs}^2\delta\frac{W}{u_{fs}}\int_0^1\frac{u}{u_{fs}}d\left(\frac{y}{\delta}\right) \qquad (8)$$

With the usual one seventh power profile, $u/u_{fs} = (y/\delta)^{1/7}$ we get

$$\frac{d\delta}{dx} + \frac{36}{7}\frac{W}{u_{fs}}\frac{\delta}{L} = 0.231\left(\frac{u_{fs}}{\nu}\right)^{-\frac{1}{4}}\delta^{-\frac{1}{4}} \qquad (9)$$

With the substitution $\Delta = \delta^{5/4}$ and the boundary condition $\delta = 0$ at $x = 0$ the solution becomes, for W constant

$$\Delta = \frac{bL}{w}\left(1 - e^{-c\eta}\right) \qquad (10)$$

where

$$c = 45\,W\,u_{fs}/7 \text{ and } \eta = x/L$$

in the limit as W approaches zero the standard flat plate boundary layer solution follows from Eq. (10) as $\Delta_{std} = bx$. Notice that for constant W the boundary layer thickness approaches a constant value as x gets large.

We have generated a solution for the boundary layer thickness and via the Blausius relationship, the wall shear. Further, the heat transfer at a point is proportional to the wall shear stress. Our solution, however, assumes that the vorticity is constant as generated in the first $x = L$ past the hole centerlines. While this could be valid if shear were neglected, our solution is applied at the plate where wall shear opposes the motion and reduces the vortex strength. The spreading of the vortex vertically as it goes downstream would also reduce the velocities associated with it.

In order to maintain mathematical simplicity and with hints from the measured data, we will assume for x/L greater than 2

$$W/u_{fs} = W_2/(u_{fs}(1+(\eta-2))) \qquad (11)$$

This function decays from $W/W_2 = 1$ at x/L = 2 to a ratio of 1 to 60 at x/L of 61 effectively reducing W to near zero at large x/L values. W_2 is determined from Eq. (9) at x/L of 2. The solution of Eq. (8) with Eq. (10) for W is

$$\frac{\Delta}{\Delta_{std}} = \left(\frac{D - D^{-w}}{1+w} + \frac{1 - e^{-2w}}{w}D^{-w}\right) \qquad (12)$$

where

$$D = (1+(\eta-2)), \quad \text{the denominator of Eq. (11).}$$

At the midline, the ratio of heat transfer with no film cooling to the heat transfer with dominance of an axial vortex (dominance only with low FST values) is

$$h/h_{nofc} = (\Delta/\Delta_{std})^{-1/5} \qquad (13)$$

This result used with Eq. (12) is compared to the measured heat transfer data to get a vorticity velocity W_2 at various blowing ratios. This is essentially picking A_s, k, and γ as functions of blowing ratio. The empirically selected values are

M	W_2
0.7	0.10
1.0	0.18
1.5	0.21

Table 1. Empirical Input Based on Heat Transfer
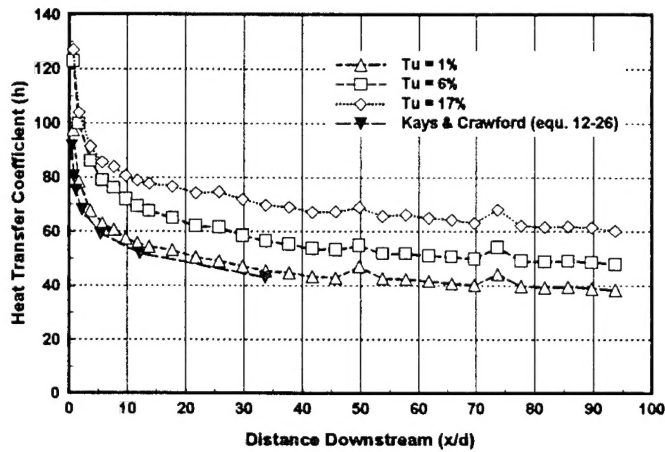
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RESULTS

Figure 3. No Film Cooling Heat Transfer Coefficient

Heat transfer coefficients are presented in Fig. 3 for no film cooling flow but three levels of FST. We see that there is a clear effect of FST with h increasing with increasing FST levels. It was also observed, but not shown here, there was virtually no effect of z location. The low FST data was within a few percent of standard results. These data are the base or standard heat transfer coefficients that are the comparison basis for the film cooling cases.

Figure 4a. Film Cooling Heat Transfer with 1% FST, M = 0.7

The low FST with film cooling heat transfer coefficients are shown in Fig. 4. at the limiting blowing ratios. The film cooling flow with a temperature between the free stream temperature and the plate temperature is designed to lower the h from the flat plate conditions shown in Fig. 3. We see that it is not only ineffective but that it raises the h at z locations near the midline for all axial locations. This heat transfer augmentation is predicted by the analytical development. This effect has long been recognized as shown in Jones and Forth (1986).
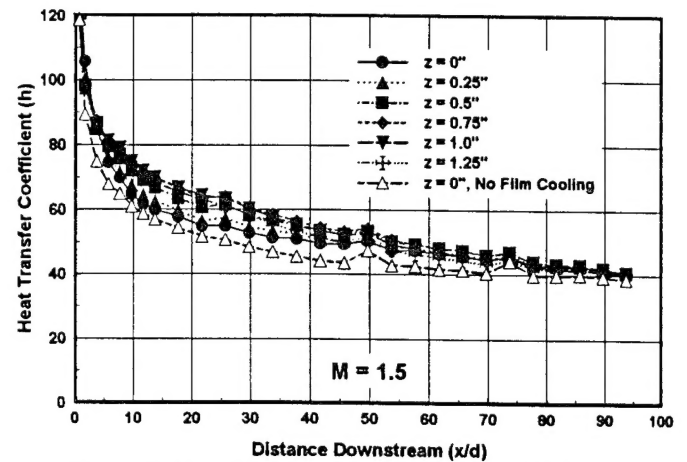

Figure 4b. Film Cooling Heat Transfer with 1% FST, M = 1.5

Figure 5 shows the same data reduced with the addition of effectiveness data both published and unpublished from the work of Bons et al. (1994). This heat transfer coefficient based on temperature difference from adiabatic wall conditions shows increased h on the midline between the holes at x/d > 10 for low FST.
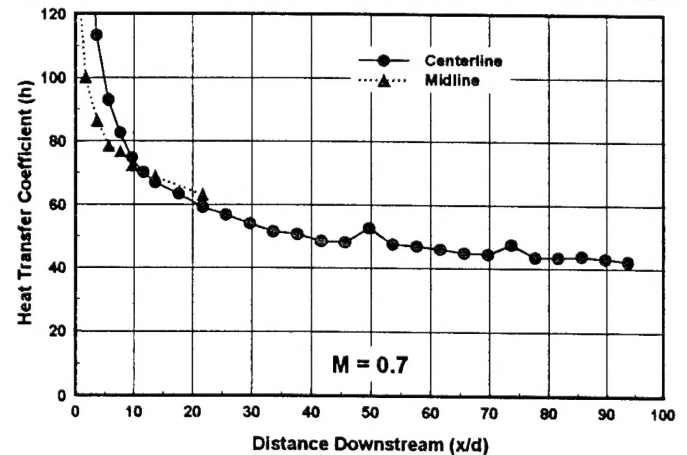

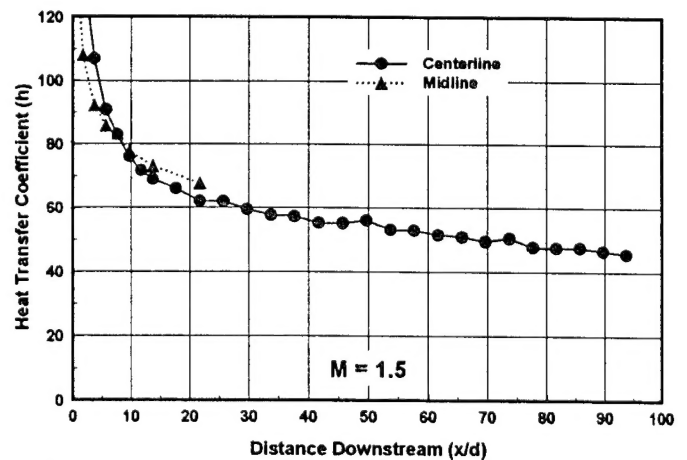Figure 5a. Adiabatic Wall Heat Transfer with 1% FST, M = 0.7

Figure 5b. Adiabatic Wall Heat Transfer with 1% FST, M = 1.5

4

Seventeen percent FST with film cooling heat transfer coefficients are shown in Fig. 6. Here the film cooling is doing the job it is designed to do, that is reduce the heat transfer coefficient. Some influence of the axial vortices is seen at x/d of 10 but the augmentation disappears as the cooling film spreads in the z direction with increasing x.
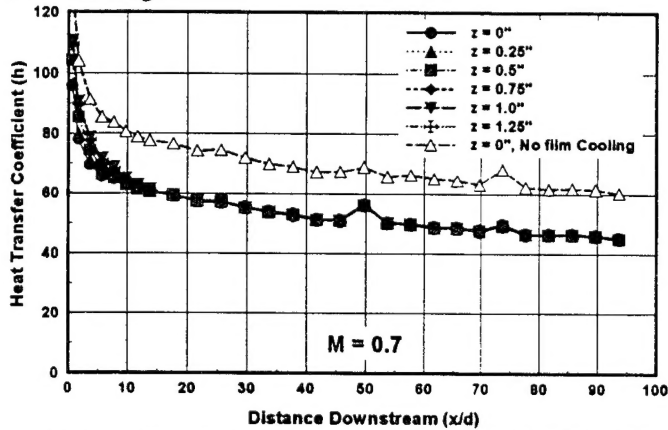

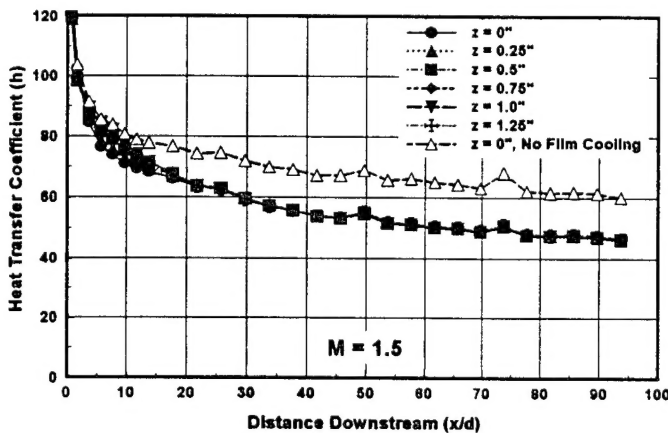Figure 6a. Film Cooling Heat Transfer with 17% FST, M = 0.7

Figure 6b. Film Cooling Heat Transfer with 17% FST, M = 1.5

This increased spread is associated with the mixing effect of the FST and is shown in Schauer and Bons (1994). The level of heat transfer is higher because the baseline from Fig. 3 is higher but the percent reduction in heat transfer has improved as FST has increased. Figure 7. shows the same data, again reduced with the addition of effectiveness data both published, and unpublished, from Bons et al. (1994). This heat transfer coefficient based on temperature difference from adiabatic wall conditions now shows a desirable lack of dependence on z at high levels of FST as does Fig. 6.

In a range of 0 < x/d < 30 the heat transfer was less for the M = 0.7 blowing ratio. For the Tu = 1% case the M = 0.7 blowing ratio produced the strongest axial vortices according to the analysis and consequently poorer results at x/d values larger than 30. The mixing in both the higher FST cases was enough that little effect of blowing ratio could be observed for x/d values greater than 30. Figures 5 and 7 show only a small effect of either M or FST level, particularly on the film cooling hole centerline.
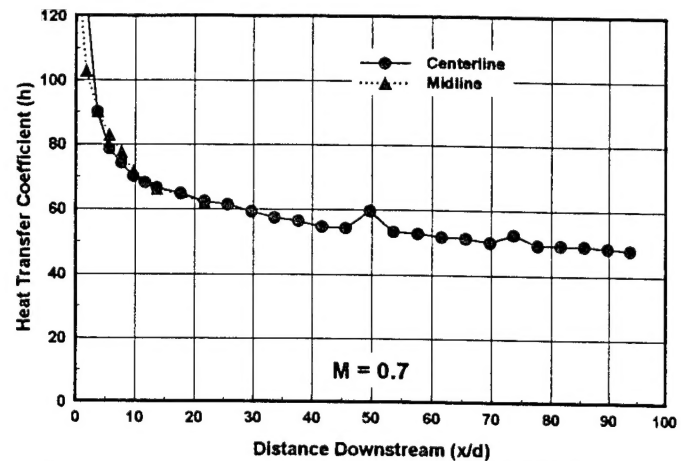

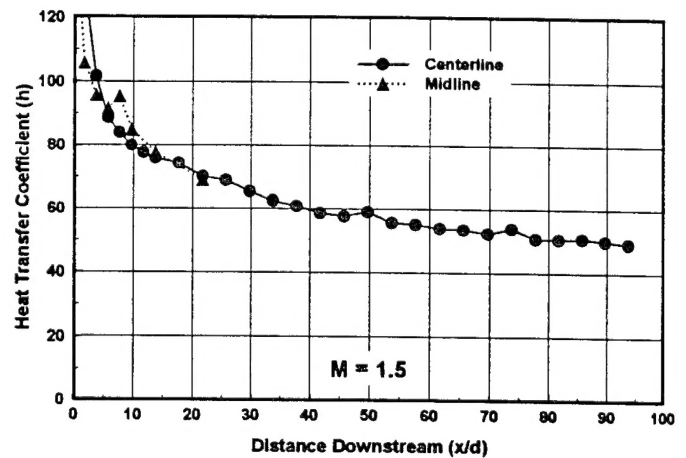Figure 7a, Adiabatic Wall Heat Transfer with 17% FST, M = 0.7

Figure 7b. Adiabatic Wall Heat Transfer with 17% FST, M = 1.5

The analytical predictions are compared to the low FST heat transfer data in Figure 8. The magnitude of the change from the no film cooling case is predicted correctly for the M = 0.7 blowing. The prediction is nearly within experimental uncertainty at M = 1.0 and also at M = 1.5 for x/d values less than 45.
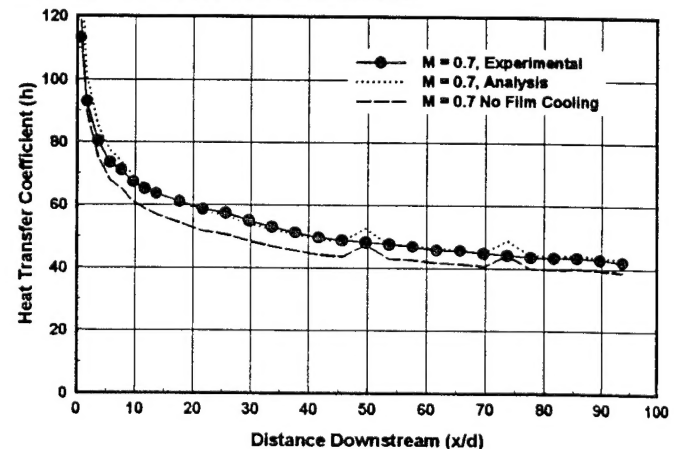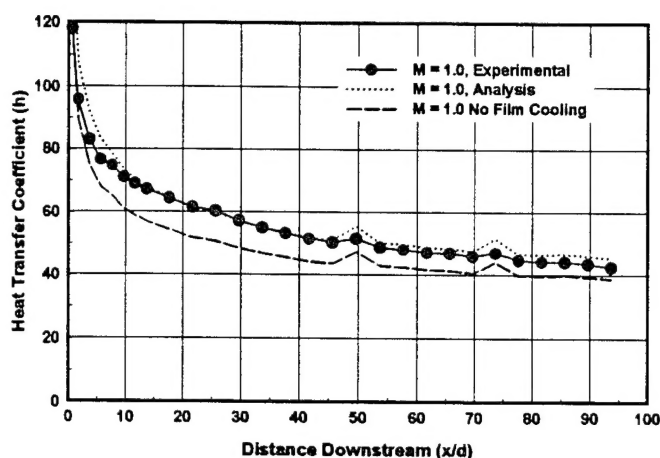

Figure 8a. Comparison of Analysis Data at 1% FST, M = 0.7

Figure 8b. Comparison of Analysis Data at 1% FST, M = 1.0

The analytical model over predicts the heat transfer augmentation at larger x/d's for M = 1.5. The prediction at M = 1.5 could be improved if the vortex decay rate was changed in Eq. (11).

The blips in the analytical results are because the deviation from the standard flat plate found analytically in Eq. (13) were applied to the no film cooling data rather than result s derived from Kays and Crawford (1980) Eq. (12-26). As seen in Fig. 3 the no film cooling data differs by only a few percent from the Kays and Crawford (1980) equation for this case of low FST.

The blips in the data at x/d of 50 and 74 are the result of two thermocouples partially detached from the heated film surface and are bad data.

CONCLUSIONS

1. Film cooling with aligned holes is only detrimental at low levels of FST. The axial vortices generated augment the heat transfer more than the shielding effects reduce it. Designers of film cooling configurations must be careful using data generated at low FST levels since mixing will be minimal and the potential for missing the effects seen in an actual high FST application are significant.

2. The undesirable heat transfer augmentation at low levels of FST is consistent with the predictions of the effects of an axial vortex generated by the whirling of the main stream flow under the film cooling jets.

3. Heat transfer between the free stream and the wall with film cooling will be minimized at an optimum level of FST. Increasing FST has two effects. The first tends to reduce heat transfer because of the increased mixing. The second tends to increase heat transfer because of the increases in the base line flat plate heat transfer as shown in Fig. 3.

4. With FST at 17%, the heat transfer coefficient on either a T_w-T_{fs} basis or T_w-T_{aw} basis is reduced about 20% for x/d greater than 20.

5. Aligned film cooling holes can produce heat transfer uniform in the transverse direction at all but very low values of FST for distances from the holes of larger than about twenty film cooling hole diameters downstream.

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